**ENGINEERING ANALYSIS AND DEVELOPMENT OF
AN ADVANCED TECHNOLOGY LOW COST
DRY COOLING TOWER HEAT TRANSFER SURFACE**

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by

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1.0 INTRODUCTION AND SUMMARY

1.1 Objectives and Scopy of Study

The overall objectives of this program are to delineate and evaluate requirements for low cost dry cooling systems, to establish conceptual and preliminary designs of advanced concept dry cooling systems, and to develop the technology required to demonstrate these concepts. The specific objectives of this initial study are "to establish the preliminary technological and relative cost advantages of an advanced concept fin tube and cooling coil geometry for use in forced or induced draft dry cooling towers for large electric power plants." Since the major costs of dry cooling systems occur in the fabrication of cooling coil elements and their headering systems, the major effort of this initial study was concentrated on investigating conceptual designs of the coil geometry and identifying relative cost advantages. Although a complete optimization of this concept was beyond the scope of this study, significant performance and cost advantages were clearly demonstrated. Also, further investigations and developments are recommended to determine and demonstrate the full advantages available with the advanced concept fin tubes which are described in detail in Section 2.0 of this study.

The method of analysis and the assumptions used in this study are described in detail in Section 3.0 of this report. To initiate this study, a review of typical requirements for a 1000 MWe forced draft dry cooling system was made and a selection of requirements for an "average plant cycle" was established. This procedure substantially reduced the scope of the required investigations, but more importantly, it provided the means by which this investigation could concentrate on the more significant evaluation of the relative performance and cost advantages available with the use of the advanced concept integral fin tubes and cooling coil geometry. It is expected that the magnitude of the relative performance and cost advantages, established herein, will remain reasonably constant with variations in the power plant cycle requirements. Investigations performed and reported herein, include evaluation of two basic types of integral fin tube configurations, i.e., single and multi-port tubes; variations in the geometry and size of the fin and tube; orientation, size and installation considerations of the cooling coil arrangement; air and water

pumping requirements; fan size and cooling coil dependence; tube drain time capabilities; tube to tubesheet joining requirements; and cooling coil fabrication considerations. These results are reported in detail in Sections 4.0, 5.0 and 6.0. Results of an investigation to compare the performance and costs of this advanced concept integral fin tubing with commercially available fin tubing are given in Section 7.0.

1.2 Integral Fin Tube Description

Curtiss-Wright has been involved in the development of high performance fin tube heat exchangers since 1950. During recent years, this effort has been directed toward the development of integral fin tubing as described in Section 2.0. In this study, two types of aluminum integral fin tubes were evaluated. One type, a single port tube, essentially is a round tube with integral fins located on opposite sides of the tube. These individual tubes are installed in a stacked assembly providing a geometry similar to a multi-port tube. Its evaluation is given in Section 4.0. The second type is a multi-port tube in which the integral fins are located on opposite sides of a flat or oval type tube with internal webs or struts. Its evaluation is given in Section 5.0. Numerous experimental performance investigations at Curtiss-Wright have confirmed the performance advantages compared to commercially available fin tubing.

The integral fins described and evaluated herein, are fabricated by a unique manufacturing process, namely, machining a row of fins from the surface of a pre-formed extrusion using a modified high speed punch press. This patented process yields a low cost high performance integral fin tube. The low cost is achieved by use of low cost billet material, high speed automated machining and lack of scrap formation in the machining process. The high performance is achieved through the use of an integral fin and tube, and by the particular design of the fin and tube geometry. The method of fabrication provides for an infinite variation potential in fin and tube geometry. Due to the limited scope of this study, only aluminum integral fin tubes were considered herein. The use of steel liners, or the use of other materials for the integral fin tubes were not considered in this study.

1.3 Results and Conclusions

Results obtained and reported herein show that the multi-port integral fin tube is clearly superior in capital and operating costs when compared to single port integral fin tubes or commercially available fin tubes in dry cooling applications. Comparative capital and operating costs are given in Section 7.0 for multi-port integral fin tubes having tube thickness from .50 to 1.25 inches vs. several commercially available round fin tubes for a range of tube diameters from .50 to 1.25 inches. For example, using the multi-port fin tubes vs. commercially available extruded fin tubes in the specified 1000 MWe size plant, results in potential cost savings of up to \$1.7 million for the tubing capital costs; up to \$2.4 million in tube joints; up to \$6,000 per each day of operation based on 10 Mill power costs; and up to \$0.5 million in fan system costs, i.e., fans, gear boxes, motors and inlet bells. Unfortunately, the above maximum cost savings do not all occur at the same tube thickness, and a further optimization is required to establish this optimum tube thickness. A description of the multi-port tube geometry used in this evaluation is given in Figure 7-1. Additional potential cost savings due to reduced numbers of structures, control systems, piping, valves, actuators, land use, etc., would also occur, but were not evaluated in this limited study.

Results presented herein also illustrate a unique type of cooling coil geometry that may provide additional cost savings with the use of the multi-port tube. This is shown in Figure 6-3 in which the cooling module is composed of a number of cooling coils which are fabricated in a serpentine geometry. It is expected that this geometry would enable use of fin tube lengths of up to 200 feet length in each cooling coil. Since the number of tube joints is inversely proportional to the tube length, this could further reduce the number, and hence, cost of tube joints by a factor of 4 to 7. Each of these cooling coils would have a single inlet and outlet fitting. Such a geometry is currently employed with this type of multi-port tubing, on a smaller size scale, in a number of automotive airconditioning condensers.

1.4 Recommendations

Although substantial cost savings were shown to be available in these preliminary investigations, the complete optimization of the integral fin and tube geometry and the cooling coil geometry was not established in this study. It is recommended that the required additional parametric evaluations be performed and the necessary cost trade-offs be evaluated such that optimum geometries can be ascertained. Also, although preliminary conceptual designs were established such that some of the significant cost items could be delineated and evaluated, additional preliminary design analyses are recommended to delineate and resolve the numerous cooling coil installation problem areas such that a demonstrator module could then be designed and fabricated. It is further recommended, that the second phase of this program be authorized to initiate the proposed fin and tube development and experimental evaluations which are necessary prior to the design and fabrication of a demonstrator module. Finally, it is recommended that a sub-scale module of a complete fan and cooling coil system be designed and fabricated to demonstrate the required performance and operational capability.

1.5 Acknowledgments

Curtiss-Wright Corporation is pleased to submit this report in support of the current ERDA development program on Advanced Concept Dry Cooling Towers for Electric Power Plants. This report is based on analytical results obtained from the first part of a proposed two part engineering study per ERDA contract number E(11-1)-2774. It is our hope that the results presented herein will justify the authorization to proceed with the second part of this study. We believe that the results presented herein will contribute significantly in helping ERDA to accomplish their program objectives.

This program was performed under the direction of W. F. Savage and T. Beresovski from ERDA, Reactor Research and Development Division. Major Curtiss-Wright contributors included: R. J. Raco, L. R. Cox, W. R. Litke, and R. J. Haberski.

2.0 DESCRIPTION OF C-W INTEGRAL FIN TUBES

Curtiss-Wright has been actively and continuously engaged in the design, development and fabrication of high performance heat exchanger equipment since 1950. Although the early effort was directed toward military applications, including liquid metal regenerators, more recent effort has been directed toward steam generators and various gas cooling applications. In particular, recent effort has been directed toward the development of high performance and low cost integral fin tube heat transfer elements for application in high volume production heat exchanger equipment. These heat transfer elements are fabricated by a unique manufacturing process, namely, by machining the fins from the surface of a pre-formed extrusion. This patented process is accomplished on a modified high speed punch press by essentially lifting a chip from the tube surface to form the fin without creating any scrap material. This process is applicable to forming integral fins on round tubes, Figure 2-1, single port flat tubes, Figure 2-2, and multi-port flat tubes, Figure 2-3. For the purpose of this study, only aluminum integral fin tubes were considered.

The low cost advantages of the C-W integral fin tube is achieved by the use of low cost billet material, high speed automated machining, and tube and fin fabrication without any significant scrap material. Since the extrusion process and the fin machining process are continuous operations, the length of the resultant fin tubing is only restricted by the available fabrication floor space. Therefore, fin tube lengths of up to 200 feet long were considered for evaluation purposes in this study. Since these integral fin tubes are machined from an extrusion, the tube wall thicknesses are normally determined by extrusion capability. Current production practice has resulted in minimum wall thicknesses of 0.030 inches while development has shown that lower thicknesses are possible. Similarly, fin thicknesses, fin heights and fin densities have been developed which are very competitive with other types of available fin tubing.

Since the integral fin tubes are fabricated from a pre-formed extrusion, the fin and tube geometric variations are infinite and are limited in size only by the capacity of the extrusion press. Therefore, tube designs are possible

with various geometry internal fins, various size, number and shape of internal ports or passages, and various shapes and size of the external tube geometry. The fin width in the gas flow direction can be made from a minimum of about .100 inches to a maximum equal to the tube width or depth. The fin width can be made uniform, varied or sized in any manner to obtain optimum performance. Since the fins are machined from the tube surface, considerable flexibility exists with respect to fin density, fin thickness and fin height. Also, since the fins are machined from the tube surface, the fin size, shape, height, pitch and thickness are independent of the tube thickness. Each of the above variables can be independently controlled such that the integral fin tube can be fully optimized based on performance and/or fabrication cost objectives.

Numerous experimental performance evaluations have been performed at C-W with these integral fin surfaces. Results have demonstrated superior performance compared to known and available round fin tube geometries. This is considered to be due to the following:

1. The integral fin concept which provides unrestricted heat transfer to the tube.
2. The size, number and shape of the fins which can be varied over a wide range and optimized for specific requirements.
3. The type, shape and frequency of fin interruptions or serrations which can be varied over a wide range to optimize performance for specific requirements. These fin interruptions inhibit fin boundary layer build-up and increase localized air turbulence resulting in improved performance compared to continuous fins.
4. The type of tube geometry which can be varied over a wide range to provide internal fin surfaces which can be optimized to satisfy specific requirements.

Although the integral fin tubes described above can be fabricated on round tubes or flat tubes, experience has shown that the flat type of integral fin tube assemblies of Figures 2-2 and 2-3 have consistently demonstrated superior

performance, i.e., less pumping power requirement, and lower fabrication costs compared to the round tube integral fin assemblies of Figure 2-1. Results presented later in this report, Section 7.0, confirm that these advantages also apply to other types of commercially available round fin tube geometries. Therefore, this study pertains only to the flat tube geometries consisting of single port or multi-port tubes. A description of these tubes is further described below:

2.1 Single Port Integral Fin Tubes

The single port tubes are essentially round tubes having integral fins machined on only two sides such that an assembly of separate tubes is made to simulate a multi-port or flat tube arrangement. Figure 2-2 shows a typical single port tube and a modular assembly of single port tubes. The tube to tubesheet joint with this tube is a conventional assembly consisting of a drilled tubesheet with conventional tube expansion. Seal welds would be used on high pressure systems, but are not considered necessary for the type of application considered in this study. Since this tube geometry consists of a round inner diameter, it has the advantage for potential use as a bi-metallic tube, i.e., it can be used with a steel tube liner for those applications wherein aluminum is incompatible with the tube fluid. Bi-metallic integral fin tubes or other fin tube materials were not evaluated in this study.

Fin shelf widths may be readily sized to provide a wide ratio of fin surface area to tube area. In Section 4.0 of this report, fin shelf width to tube thickness ratios of 1.2 to 4.0 were evaluated. Typical values for a .50 I.D. tube with .80 fin height, 0.011 fin thickness, 10 fins per inch, and 4.0 ratio of fin shelf to tube thickness results in fin-side surface area to tube-side surface area ratio of 50. To simulate a multi-port tube geometry, single port tubes are designed with an interlocking system at the tube sides and arranged in a stacked assembly such as shown in Figure 2-2. For wide tube shelf widths, i.e., tube width to thickness ratio greater than 2, a strut is required to support the fin shelf. This strut is made only thick enough to support the fin machining process. It also serves the function of providing additional tube interlocking or positioning guides.

2.2 Multi-Port Integral Fin Tubes

Multi-port tubes are flat or oval type tubes having integral fins on the two parallel sides. These external fins are machined simultaneously on both sides of the tube. The extruded tube is formed with a number of internal webs which provide the necessary support for the finning process and also provides the required structural strength to satisfy the design tube side pressure. These webs also increase the fin tube performance since they act as internal fins. Figure 2-3 shows a sample of typical multi-port integral fin tube geometries.

A major application currently using this type of integral fin tubes in a multi-port configuration is in the Freon condenser for a number of automotive air conditioning systems. This type of core is shown in Figure 2-4. This assembly is made from a single tube which is bent to 180 degrees such that the fin tips of one tube section just touches the fin tips of the next tube section resulting in 14 equivalent tube lengths. This is made possible due to the unusual flexibility of this type of tubing which allows it to be bent to small radii without significant distortion of the tube walls. A simple retainer frame provides the required structural support and mounting provisions. Although the face area of this core measures only 13 x 21 inches, the tube length is 32 feet. This results in substantial economic savings due to the need for only 2 tube joints or fittings, i.e., inlet and outlet fittings, rather than the 28 tube joints that would have been required for these 14 tube sections if tube bending were not used. It is an objective of this study program to evaluate the feasibility and economics of this concept, i.e., using large cooling cores made up of multi-port tubes of lengths up to 200 feet. With such a system applied to dry cooling towers, cost savings for tube joint fabrication, tube fabrication and core handling would be achieved.

Since the multi-port tube ends are oval in shape, conventional drilled, reamed and tube expansion type of tube to tubesheet joints cannot be used. Instead oval slots must be formed in the tubesheet. This can be accomplished by various machining processes, but for this application a die punched operation is considered to be satisfactory and very economical. A construction of tube

to tubesheet assembly, which is suitable for this application using multi-port tubes, is shown in Figure 2-5. This assembly consists of a tubesheet, a seal plate and a retainer or press plate. The tubesheet provides the structural element, the seal plate provides the water to air seal by means of an elastomeric O-ring, and the press plate is used to retain the O-ring in position. To provide the required surface finish on the seal plate, the punched slot operation is followed by a broaching operation. This broaching is not required on the tubesheet or retainer plate. The remaining assembly of tubesheet closures and mainfolds would be the same as would be used with round tubes. An evaluation of multi-port tubes for this application is given in Section 5.0.

Axial Fin-Tube

Rotary Fin-Tube

Figure 2-1 Integral Fin and Round Tube Heat Exchanger Elements

Figure 2-2 Single Port Integral Fin Tube Heat Exchanger Elements and Module Assembly

Figure 2-3 Various Types Of Multi-Port Integral Fin Tube Heat Exchanger Elements

Figure 2-4 Integral Fin Tube Automotive Air Conditioning Condenser Assembly

Figure 2-5 Multi-Port Tube to Tube Sheet Assembly

3.0 ASSUMPTIONS AND METHOD OF ANALYSIS

Since the function of a cooling coil in a dry cooling tower is to use atmospheric air to cool the condenser water of a power plant, the existing design techniques of Reference 1 for air-to-water heat exchangers were used in this design and analysis. Because of the large number of variables to be considered in the evaluation and analysis leading to the cooling coil design, it was advantageous to proceed as follows: (1) C-W integral finned tubing having single ports and multi-ports were evaluated from a geometric and a cooling coil performance point of view, (2) using the results of Step 1 above, single port tubes were compared with multi-port tubes; here it was found that multi-port tubes were superior, and (3) establish conceptual cooling coil designs containing one or more cooling coils incorporated into a fan module. The final step of completely optimizing the coil designs and incorporating the optimized modules into a low cost dry cooling tower design was beyond the scope of this work. However, in all steps of the above procedure, the evaluation and analysis were always concerned with the ultimate goal of obtaining the lowest possible cost, i.e., the sum of capital and operating costs. Other factors that were considered include the fan/heat exchanger relationship and the draining time as discussed below.

3.1 Power Plant Requirements and Average Cycle Concept

In order to initiate this study, a range of typical requirements for a 1000 MWe induced draft cooling system was established with ERDA. Examination of these requirements led to a selection of an "average cycle" which was used in this study in order to investigate the cooling coil characteristics, size, performance and cost factors. The selection of an average cycle also provided a basis for investigation of installation requirements. For the purposes of this study, i.e., to evaluate and delineate relative performance and cost advantages of the C-W integral fin tube, the concept of using an average cycle provides a direct means of preliminary evaluation and comparison without requiring an extensive investigation throughout the entire range of power plant variables. A listing of the established range of power plant variables along with the selected average cycle conditions are summarized below.

<u>Power Plant Variable</u>		<u>Typical Range for Power Plants</u>	<u>Value for Selected Average Cycle</u>
1.	ITD	40 to 70°F	55°F
2.	Water Range	0.1 to 0.5 ITD	16.5°F
3.	Fan Power		Calculation method explained in Section 3.2
	Fossil	1.0 to 2.5% of plant output (13,400 to 33,500 HP).	
	Nuclear	1.5 to 3.0% of plant output (20,100 to 40,200 HP).	
4.	Pump Power (Condenser Piping and Cooling Coil)		Calculation method explained in Section 3.2
	Fossil	0.5 to 1.0% of plant output (6,700 to 13,400 HP).	
	Nuclear	0.8 to 1.5% of plant output (10,720 to 20,100 HP).	
5.	Heat Rejection (at 3.5 in. of mercury condenser pressure) for 1,000 MWe (1.341 X 10 ⁶ HP)		5.75 X 10 ⁹ BTU/HR
	Fossil	4.8 X 10 ⁹ BTU/HR	
	Nuclear	6.7 X 10 ⁹ BTU/HR	
6.	Energy Charge	10 MILLS/KW-HR	10 MILLS/KW-HR
7.	Fan Data	From Reference 2	From Reference 2
8.	Drain or Fill Time	30 to 90 seconds	30 to 90 seconds
9.	Bundle Size	Limited to shipping	Limited to shipping

3.2 Basic Assumptions and Specifications

The three items of prime importance in evaluating the performance of the cooling coils used in forced or induced draft dry cooling system are: (1) the heat rejection rate, (2) the water pumping power, and (3) the fan power. The procedure for calculating the heat rejection rate and fan power is explained in detail in Reference 1. The procedure for calculating the water pumping power is given in detail in Reference 3. The defining equations (see Section 9 for list of symbols) used are:

1. Heat transfer rate:

$$q = E(Wc_p)_a (T_w - T_a)_{in} = (Wc_p)_a (T_{out} - T_{in})_a = (Wc_p)_w (T_{in} - T_{out})_w$$

2. Water pumping power:

$$P_w = \Delta p_w W_w / \rho_w$$

3. Fan power:

$$P_a = \Delta p_a W_a / \rho_a$$

For cooling coil applications of practical importance, the effectiveness (E) varies from 0.3 to 0.7 (References 1 and 4). Therefore, the definition of effectiveness, from Reference 1, gives:

$$E = (T_{out} - T_{in})_{air} / (T_{water} - T_{air})_{in}.$$

For this study the performance of the cooling coils were analyzed as the effectiveness varied from 0.35 to 0.65 (Sections 4.2 and 5.2). Here the air inlet temperature used was as specified for the average cycle and the water inlet temperature was equal to the steam temperature in the condenser so that the air outlet temperature was calculated from the effectiveness equation. Using the values of the water inlet and outlet temperature as specified for the average cycle, the total water flow rate (W_w) was calculated from equation (1) above. Knowing the total water flow and the coil geometry, the water pressure drop (Δp_w) was calculated by the method of Reference 3 as:

$$\Delta p_w = (\Delta p_w)_{\text{friction}} + (\Delta p_w)_{\text{bend}}$$

$$= \left(f \frac{L_T}{d_H} \frac{G^2}{2\rho} \right)_w + \left(\frac{N_B K_B}{d_H} \frac{G^2}{2\rho} \right)_w$$

The heat transfer rate and the fan power of the Curtiss-Wright integral finned tubing as calculated by the method of Reference 1 require the use of the Colburn J factor and the friction F factor. For this analysis, the J and F factors for the Curtiss-Wright integral fins as experimentally determined by Reference 5 were used. From experimental data obtained by C-W for the C-W integral finned tubing, it is known that small changes in the fin geometry produce small changes in the F and J factors. Hence, for this study, because of the range of fin geometries analyzed, the effect of the fin geometry on the F and J factors was neglected in order to obtain a first approximation of the relative size and performance of the cooling coils. The complete optimization of a cooling coil, which is beyond this work scope, would consider the effects of all the cooling coil variables and account for the variation of the F and J factors with the fin geometry.

3.3 Sensitivity Evaluations

In this study the following fin dimension was taken as the base design dimensions:

1. Fin pitch - 10 fins per inch.
2. Fin thickness - 0.011 inches.
3. Fin height - 1.0 inches.

In this sensitivity study, the fin pitch was varied from 8 to 12 fins per inch, the fin thickness was varied from 0.006 to 0.021 inches and the fin height was varied from 0.6 to 1.0 inches in order to ascertain the effect of the fin geometry on the heat exchanger performance. Although a final optimization was not achieved in this study, a relative comparison of cooling coil size and performance was accomplished and these comparative results can be expected to remain reasonably consistent with a more complete study.

3.4 Fan/Cooling Coil Relationships

The axial flow fan characteristics used in this study to establish air pumping power and hourly costs were based on experimental performance data provided in Reference 2 for fan diameters ranging from 4 to 30 feet. These data give the relationships between fan diameter, blade pitch, number of blades, airflow, velocity pressure, horsepower and total pressure. In Reference 6 it is stated: "Velocity pressure should fall in the range of 0.15 to 0.25 in. for optimum performance." For this study, therefore, the fan exit velocity pressure was assumed constant at .2 inches of water and was further assumed to be discharged to the atmosphere and therefore was unavailable for recovery to the system. In addition, the fan tip speed was maintained constant at 12,000 feet per minute since this value is near the upper limit due to noise generation and was consistent with the values for the fan curves of Reference 2. Using the data and the above assumption for velocity pressure and fan tip speed, a generalized fan performance curve was generated giving the airflow per horsepower versus the available total head. These data apply for all fan diameters up to 30 feet as long as the number of blades is selected to yield the maximum efficiency for the required pressure level and diameter. For the analysis of cooling coil performance, the available total head given by the generalized fan performance curve must be equal to, or greater than, the air pressure drop through the cooling coil. Hence, for this study, once the air pressure drop and the air flow through the cooling coil were determined by the method of Reference 1, the generalized fan performance curve was used to obtain the required fan horsepower.

Next, it was found that at the selected velocity pressure and tip speed, the fan flow divided by the fan diameter squared was approximately 1.6 pounds per second per square foot for all fan diameters up to 30 feet. Then from preliminary layouts of practical fan/cooling coil configurations, a ratio of cooling coil frontal area to fan diameter squared of approximately 1.6 was established. Thus, by ratio of the two above results, the air mass velocity per square foot of cooling coil frontal area is one pound per second. Therefore, for this study, the air mass velocity per square foot of cooling coil frontal area was maintained constant at one pound per second during the parametric and sensitivity studies reported herein. The air pumping horsepower

obtained from this result is consistent with the results of the generalized fan performance curve. Although the use of the general fan performance curve was sufficiently accurate for this study, a complete optimization of any cooling coil analyzed would use the particular fan characteristic curve for each fan to be considered to give better accuracy. The water pumping powers were calculated using an assumed pump efficiency of 80% which is consistent with commercially available pumps.

3.5 Drain Time Considerations

Since the maximum draining time of the cooling coil was specified at 90 seconds, it was necessary to be able to predict the drain time for all coil designs analyzed. The ordinary differential equation as deduced from Reference 3 that relates the length (X) of tube drained in time (t) is given as:

$$\frac{d^2 X}{dt^2} + \frac{f}{2d_H} \left(\frac{dX}{dt} \right)^2 - gH_e = 0$$

Here, f is the friction factor and g is the acceleration due to gravity. Hence, it is seen that the drain time depends upon the tube geometry, i.e., the hydraulic diameter (d_H) and the effective head (H_e) where the effective head is defined as the ratio of the vertical tube height to the total tube length. Therefore, the effective head varies from zero (horizontal tube) to unity (vertical tube). Numerical solutions of the draining equation were evaluated for the various cooling coil configurations of Section 6.0 in order to establish the maximum allowable tube lengths.

4.0 EVALUATION OF SINGLE PORT TUBES

An evaluation of single port integral fin tubes is reported in this section while in the following section, Section 5, an evaluation of multi-port integral fin tubes is given. Also, a comparative evaluation of single port and multi-port integral fin tubes was made in Section 5 in order to select the tube geometry having the greater economic advantage for application to dry cooling tower systems. In this study, the evaluation of both single port and multi-port integral fin tubes was directed toward ascertaining the total cost which was equal to the sum of:

1. The capital cost which is related to the amount of tubing required, and
2. The operating cost which is related to the performance of the tubing.

In this analysis, it was advantageous to further divide the evaluation into geometric considerations and cooling coil performance considerations which are described below.

4.1 Effect of Tube Geometry

The effects of the single port tube geometric parameters were analyzed in order to help in the selection of the most advantageous tube geometry. The effects of the following parameters were analyzed in this study: (1) shape factor, (2) tubing cross-sectional area, (3) tubing metal volume per cooling coil volume, (4) water pressure drop, and (5) the heat transfer ratio. For ease of presentation, these parameters are discussed separately. However, it is important to note that the effects of all the above parameters were considered jointly so that their relative importance was included in the evaluation of potential cooling coil designs using single port tubes.

4.1.1 Shape Factor

The heat rejection for a cooling coil is proportional to the heat transfer rate through the tube wall per unit length of tubing which is given as:

$q' = \Delta T_m / R_m = k S \Delta T_m$. Therefore, for a given metal conductivity (k) and

metal temperature difference (ΔT_m), the heat rejection increases as the shape factor (S) increases so that from an operating cost point of view, a tube with a large shape factor is desired. However, with all other things being equal, from a capital cost point of view, a tube with a small cross-sectional area is desired. Therefore, from a total cost point of view, a tube with the maximum value of shape factor for the smallest cross-sectional area is superior. Hence, the shape factor analysis performed was directed toward obtaining the above objective as follows:

For simple geometries, the shape factor is tabulated in the literature, for example, Reference 7 gives the shape factor for a rectangle with a hole at its center (Figure 4-1) as

$$S = 2\pi / \ln \{ (2 T_W / \pi d_i) \sinh (\pi T_T / 2 T_W) \}$$

In order to determine the shape factor for the complex single port tube geometries analyzed, an experimental method which makes use of the fact that the flow of heat is analogous to the flow of electricity was used. For electricity, $q_e = \Delta V / R_e$, while, as previously stated, for heat transfer $q' = \Delta T_m / R_m$. Hence, by ratio, $S = K / R_e$. The value of K was easily determined by (1) calculating the shape factor, using the above equation for the above mentioned rectangle having its sides equal to the single port tube width and thickness, respectively, and a hole in its center equal to the I.D. of the single port tube (Figure 4-1), and (2) measuring the electrical resistance (R_e) of electrical conducting paper of relatively high resistivity that was cut into the shape for which the shape factor was calculated. Then to determine the shape factor of the single port tube, the unwanted portions were cut from the electrically conducting paper so that it had the shape of the tube cross-section and its electrical resistance was measured with an ohmmeter. Using this procedure, the shape factor was determined for single port tubes having internal diameters of 0.5, 1.0 and 1.75 inches and tube width to thickness ratios of 2, 3 and 4. The results in Figure 4-1 show that the thicker tubes have a larger shape factor than the thinner tubes and the shape factor decreases as the tube width decreases.

Hence, from a heat transfer point of view, large tubes are more efficient than small tubes. However, from Figure 4-1, it is also seen that the decrease in shape factor due to reducing the tube cross-sectional area is smaller for the small tubes than for the large tubes. In dry cooling tower applications where the metal heat transfer resistance (R_m) is smaller than the air heat transfer resistance, the change in shape factor due to the change in tube cross-section area is of secondary importance. Therefore, because of a lower capital tube cost, small tubes would be superior. The results of Figure 4-1 and the results from the evaluation of the other geometry effects were incorporated into the evaluation of the cooling coil performance using single-port tubes in order to select the single port tube giving the cooling coil having the lowest total cost.

4.1.2 Tube Cross-Sectional Area and Tube Metal Volume Per Cooling Coil Volume

The product of the tube cross-sectional area and the feet of tubing required per cooling coil gives the volume of tube material (excluding fin volume) required per cooling coil. Because the material cost is directly related to the material volume, the tube cross-sectional area gives a measure of the tubing cost per foot of tube. In this study, the tube cross-sectional area was calculated for the above defined single port tubes. Figure 4-2 shows that the smallest tubes have the smallest cross-sections, and hence, the smallest cost per foot. Further, it is seen that the cross-sectional area increases with the tube diameter, i.e. the tube thickness.

Besides desiring knowledge of the tube cross-sectional area, it is also informative to know the tube metal volume per cubic foot of cooling coil volume. This knowledge is helpful in determining the core cost for a given cooling coil volume and also is a measure of the compactness of the cooling coil. values of this parameter are plotted in Figure 4-2. It is seen that cooling coils using smaller tubes are more compact. These results were considered jointly with the results from the evaluation of the other geometry effects to establish the single port tube having the greatest economic advantages. That is, the analysis used in obtaining the results for the tube cross-sectional area and the tube metal volume per cooling coil volume was extended in order

to obtain the total amount of finned tube metal volume required per cooling coil. This analysis and the analysis of Section 4.2 for the evaluation of the cooling coil performance were incorporated into a computer program for obtaining a measure of the capital cost of the tubing.

4.1.3 Water Pressure Drop Parameter

From Reference 3 and as shown in Section 3.2, the water pressure drop per foot of tube is proportional to the mass velocity (G) squared divided by the hydraulic diameter (d_H), i.e., $\Delta p/L_T \sim G^2/d_H$. Now the mass velocity is inversely proportional to the total flow area which is equal to the area per port (A) times the total number of ports (N_p), i.e., $G \sim 1/(A N_p)$. Hence, by considering one tube row in the air flow direction, the number of ports is inversely proportional to the tube width (T_W), so that $G \sim T_W/A$. Therefore: $\Delta p/L_T \sim 1/(d_H A^2/T_W)$. By defining a water pressure drop parameter for single port tubes as: $D_S = d_H A^2/T_W^2$, it is seen that the water pressure drop parameter is inversely proportional to the water pressure drop. Values of this parameter were calculated for the same sizes of single port tubes analyzed in the previous section and plotted in Figure 4-3. Here it is seen that tubes with big holes and the smallest tube width to height ratio have the smaller water pressure drop. These results were also used in Section 5.3 to compare single port tubes with multi-port tubes.

4.1.4 Heat Transfer Resistance Ratio

From Reference 1, the heat rejection rate varies inversely with the overall heat transfer resistance of the tube (R). For a single port tube, the overall heat transfer resistance is given as the sum of the air side resistance (R_o) plus the metal resistance (R_m) plus the water side resistance (R_i).

Hence, the ratio of the water side resistance to the overall resistance is given as: $R_i/R = 1/(1 + R_o/R_i + R_m/R_i)$. From this equation the heat transfer resistance ratio becomes unity only when the water side heat transfer resistance becomes controlling, i.e., larger than the metal and air side heat transfer resistance. Hence, for dry cooling tower applications, tubes with small values of this ratio are desired. In order to calculate values of this ratio, the method of Reference 1 was used to calculate the individual heat

transfer resistances. By considering the practical range of cooling coil parameters and geometric parameters as guided by Reference 4 and the results of previous experience available at C-W, the range of this ratio varied from 0.18 for small diameter tubes to 0.28 for large diameter tubes. From this result, it was concluded that small diameter tubes would have a superior heat transfer performance. Because the previous section concluded that large diameter tubes have a lower pumping cost, the effects of all of the geometric parameters were considered jointly in the following section that evaluates cooling coils using single port tubes.

4.2 Cooling Coil Performance

The evaluations discussed in Section 4.1 were based on the effects due to the tube geometry alone. In this section, the cooling coil performance for potential cooling coil designs were analyzed in order to ascertain the effects of the heat exchanger effectiveness, tube length, tube I.D., tube width, and number of passes on the required number of tubes, total metal volume, water pumping power, fan power, total power, and the cooling coil depth. That is, for the average cycle requirements and the analysis method of Section 2, the procedure of Reference 1 was used to analyze a cooling coil using single port tubes having a fixed length, I.D., width, number of passes and effectiveness. Guided by the results of Reference 4 and previous C-W experience, preliminary studies were made for four different single port tubes with a fixed length of 20 feet, having 1, 2, 3 and 4 passes, having effectiveness from 0.35 to 0.65 and dimensions as follows: (1) I.D. of 0.52 inch with 0.72 and 2.4 inch tube widths, and (2) I.D. of 0.92 inch with 1.2 and 4.0 inch tube widths. Typical results from the preliminary portion of this study showing the effect of the effectiveness and number of passes for the 4 different single port tubes are given in Figures 4-4 and 4-5. It is seen that the 0.52 inch I.D. tube requires less power than the 0.92 inch I.D. tube, the difference between one and two passes is insignificant for the 0.52 inch I.D. tube, and that the smallest single port tube (0.50 inch I.D. and 0.72 inches wide) requires the smallest amount of power. For the subsequent portion of this study, the above procedure was extended to ascertain the effect of the tube length. Figure 4-6 shows that as the tube length increases, the number of tubes required decreases, however, the cooling coil depth, metal volume and power required

increase. Therefore, a tube length from 20 to 50 feet would require the smallest power for the single pass 0.52 inch I.D. and 0.72 inch wide single port tube. On the basis of the minimum power requirement, this tube was selected for comparison with multi-port tubes in the next section.

SHAPE FACTOR AS A FUNCTION OF SINGLE PORT TUBE GEOMETRY

Figure 4-1

SINGLE PORT TUBE GEOMETRIC PARAMETERS

Figure 4-2

WATER PRESSURE DROP PARAMETER AS A
FUNCTION OF SINGLE PORT TUBE GEOMETRY

Figure 4-3

PERFORMANCE OF ONE PASS SINGLE PORT TUBES

Figure 4-4

EFFECT OF NUMBER OF WATER PASSES ON SINGLE-PASS TUBE PERFORMANCE

Figure 4-5

COMPARISON OF SINGLE-PORT INTEGRAL FIN TUBES

4-12

Figure 4-6a

COMPARISON OF SINGLE-PORT INTEGRAL FIN TUBES

5.0 EVALUATION OF MULTI-PORT TUBES

Beside evaluating multi-port tubes, this section compares multi-port tubes with single port tubes, discusses the advantages of multi-port tubes over single port tubes and discusses the sensitivity of tube and fin parameters for multi-port tubes. The remarks made in Section 4 about the evaluation methods for single port tubes also apply to multi-port tubes, therefore, they are not repeated here.

5.1 Effect of Tube Geometry

The webs joining the tube walls of multi-port tubes not only create the water passages, but also reduce the thickness of the tube walls by decreasing the wall stress. These webs also increase the tube performance because they act as internal fins. Since the multi-port tube is made by an extrusion, it can be produced in an unlimited number of tube cross-section geometries. Because of the unlimited geometric variations, more analysis was required for the multi-port tube than the single port tube. However, these additional geometric variations allowed for a better selection of the most advantageous multi-port tube geometry.

5.1.1 Tube Cross-Sectional Area and Tube Metal Volume Per Cooling Coil Volume

The importance of these parameters and results for single port tubes were discussed in Section 4.1. Although a number of tube geometries were investigated, the results for only one typical set of tube dimensions are presented (Figure 5.1). Here it is seen that the cross-sectional area is affected mainly by the wall thickness when the web thickness is maintained constant. Also, the tube metal volume per heat exchanger volume increases as the wall thickness increases and as the hole width increases; however, it does not increase as rapidly as the tube cross-sectional area increases. This means that the multi-port tubing is very compact, and hence, leads to smaller depth cooling coils than single port tubes.

5.1.2 Water Pressure Drop Parameter

The importance of the water pressure drop parameter was discussed and defined for single port tubes in Section 4.1. Because the tube width per port for a

multi-port tube is equal to the sum of the hole width plus the web thickness, the water pressure drop parameter for multi-port tubes is defined as $D_M = d_H A^2 / (H_W + S_T)^2$. For multi-port tubes, the hole width, hole height and the port fillet radius affect the flow area (A) and the hydraulic diameter (d_H), and hence, the water pressure drop parameter. Because of this dependency, the water pressure drop parameter was able to be optimized with respect to each of these dimensions and the results are shown on Figure 5-2. All of the results in this section are based on tube ports yielding minimum water pressure drop, while results given in Section 7.0 are based on tube ports yielding minimum tube metal volume.

Using the results of Figure 5-2, a curve-fit equation was developed that related the maximum value of the water pressure drop parameter to the tube thickness, port fillet radius and hole width. This curve-fit equation was then used in the analysis of multi-port tube cooling coil designs when the smallest possible water pressure drop was desired as governed by the tube shape - tube thickness relationship. When the smallest possible metal volume was desired, then the analysis of multi-port tube cooling coil designs considered rectangular ports using a minimum port fillet radius.

5.1.3 Heat Transfer Resistance Ratio

The definition, importance and details related to this calculation were given in Section 4.1. For multi-port tubes of typical geometry, it was found that the effect of the metal heat transfer resistance is insignificant because (1) the webs act as internal fins, and (2) the webs allow for the use of a thin tube wall; both of these factors keep the metal heat transfer resistance small. Also, the effect of the ratio of the tube width to the tube thickness showed that thicker tubes have a superior heat transfer performance. From the results of Reference 4 and from prior C-W experience with cooling coils, the value of the heat transfer resistance ratio varied from 0.10 for a 1.0 inch thick tube to 0.22 for a 0.30 inch thick tube.

5.2 Multi-Port Tube Cooling Coil Performance

In Section 4, the performance of a 0.52 and a 0.92 inch I.D. single port tube was analyzed to determine the tube length requiring the lowest total power,

i.e., for a constant diameter, the tube length was optimized to give the lowest operating cost. Because multi-port tubes are not limited to circular cross-sections, they can be optimized with respect to the water passage size and shape as well as the tube length. In order to reduce the amount of calculations required to achieve an optimum design, the port size was related to the port shape as mentioned in Section 5.1.2, i.e., the multi-port tube analysis considered rectangular ports as governed by (1) the minimum metal volume, and (2) by the minimum water pressure drop. Hence, for a fixed geometry and effectiveness, the cooling coil cost optimization was dependent upon the tube thickness and the tube length as shown schematically in Figure 5-3. Parts A and B of Figure 5-3 illustrate that if all of the variables are held constant except for the tube thickness and tube length, then for the chosen port size/shape relationship, there is only one value of tube length and tube thickness that gives the lowest cooling coil cost. As discussed in Reference 8, the determination of the lowest cost becomes exceedingly tedious as the number of variables increase. Because a complete optimization which requires the consideration of all cost factors and cooling coil variables was beyond the scope of this work, the following procedure was used. The analysis first considered only the effects of the tube length and tube thickness for tube port shape having minimum water pressure drop. After the effects of these variables were ascertained, the effect of the heat exchanger effectiveness, i.e., water and air temperatures were determined. Note that when the effects of all the variables are included, then a complete optimization would be obtained.

5.2.1 Minimum Power Analysis

This analysis began by (1) determining the effect of the tube length on the total power cost when the tube thickness was held constant, i.e., planes parallel to plane B of Figure 5-3C; and (2) determining the effect of the tube thickness on the total power cost when the tube length was held constant, i.e., planes parallel to plane A of Figure 5-3C. As shown by Figure 5-3C, for the constant tube length illustrated (i.e., plane A), point 1 represents the optimum tube thickness giving the lowest power cost. However, for the constant tube thickness illustrated (i.e., plane B), point 2 represents the optimum tube length giving the lowest power cost. Hence, from Figure 5-3C, it is obvious that the optimum tube thickness for a given tube length (point 1)

is not the same as the tube thickness at which the optimum tube length occurs (point 2). Results in this section are based on planes parallel to plane A in Figure 5-3C, while results given in Section 7.0 are based on planes parallel to plane B in Figure 5-3C.

By following the procedure for the analysis outlined above, Figure 5-4 was obtained when the cooling coil performance of potential cooling coil designs were evaluated according to the method of Section 4, for tube ports yielding the minimum water pressure loss, tube lengths from 10 to 100 feet and for tube thicknesses from 0.3 to 1.0 inches. The same fin geometry as that of Section 4 was used, i.e., 10 fins per inch 0.011 inch fin thickness, and 1.0 inch fin height. From Figure 5-4, it is seen that the minimum power is affected more strongly by the tube thickness than by the tube length. That is, in order to obtain the minimum power at a given tube length, the tube thickness must be increased slower (curve A) than the increase in the tube length for a given tube thickness (curve B).

Next, the analysis was extended to include the effects of the heat exchanger effectiveness. Multi-port tubes with the fin geometry given in Section 4.2 were evaluated for tube lengths from 20 to 200 feet long. This range of lengths was considered because it is the range of practical interest and provides for the use of the maximum tube lengths that might be considered, as discussed in Section 6.0. Typical results for a 200 foot long multi-port tube are shown in Figure 5-5. Here it is seen that a minimum power occurs with respect to the tube thickness and effectiveness, i.e., about 1.15 inches thick and 0.53 effectiveness. Similarly, the minimum power with respect to tube thickness and effectiveness was determined for tubes from 20 to 200 feet long and results are summarized in Figure 5-6. Results show that in order to provide a cooling coil having the minimum total power, the tube thickness, number of tubes, depth and metal volume must be functions of the tube length as shown in Figure 5-6.

5.3 Comparison of Multi-Port Tubes with Single Port Tubes

Based on the evaluations of Sections 4 and 5, it is seen that multi-port tubes are superior to single port tubes as described below.

1. Tube cross-sectional area and tube metal volume per heat exchanger volume: Comparing Figure 4-2 with Figure 5-1, it is seen that the multi-port tubes have a smaller tube cross-sectional area per port than single port tubes for the same tube thickness. Therefore, multi-port tubes cost less per port per foot than equivalent single port tubes. Also, it is seen that the multi-port tubes have a smaller tube metal volume per heat exchanger volume. Therefore, any given size heat exchanger will require less metal volume when multi-port tubes are used than when single port tubes are used.
2. Water pressure drop parameter: Comparing Figures 4-3 and 5-2, it is seen that for the same tube thickness the multi-port tubes have a larger value of the water pressure drop parameter, and hence, a lower water pressure drop and water pumping cost than the single port tubes.
3. Heat transfer resistance ratio: Referring to Sections 4.1.4 and 5.1.3 it is seen that the ratio of water side resistance to overall resistance is lower for the multi-port tubes than for the single port tubes and therefore, they have a more favorable thermal performance and operating cost than single port tubes.
4. Heat exchanger performance (Figure 5-6):
 - (a) The multi-port tube heat exchanger requires from 12% to 28% less operating power than a single port tube heat exchanger.
 - (b) Multi-port tubes require fewer tubes, and hence, fewer tubesheet joints than single port tubes by a factor of 10.
 - (c) For tube lengths greater than 40 feet, the depth of the multi-port tube heat exchanger can be up to 4 inches less than the depth of the single port tube heat exchanger.
 - (d) The multi-port heat exchanger requires 35% less tubing metal than a single port tube heat exchanger.
 - (e) The superiority of multi-port tubes for items (a) to (d) above increases with tube length, hence, it becomes even more advantageous to use multi-port tubes rather than single port tubes for compact cooling coils as described in Section 6.0.

5.4 Sensitivity Evaluation of Multi-Port Tubes

Because of the superiority of multi-port tubes, sensitivity studies related to the fins per inch, fin length and fin thickness were evaluated for multi-port tubes only. A straight multi-port tube with a 40 foot length (as based on a maximum truck shipping length) with 10 fins per inch, 0.011 inch fin thickness, 1.0 inch fin length, with minimum water pressure drop ports, 0.38 inches tube thickness and a 0.58 effectiveness was evaluated according to the methods of Section 4. Note that the tube thickness and effectiveness were as required by Figure 5-6 to give the minimum power for the 40 foot tube length.

For the tube geometry given above, the performance of the heat exchanger was analyzed as the fin height varied from 0.6 to 1.0 inches and the fin pitch varied from 8 to 12 fins per inch for a constant fin thickness of 0.011 inch (Figure 5-7). It is seen that the power required has a minimum at a fin height of about 0.8 inch. In order to further minimize the power while ascertaining the effect of the fin thickness, the performance of the heat exchanger was then analyzed as the fin thickness varied from 0.006 to 0.021 inches and the fin pitch varied from 8 to 12 fins per inch for a constant fin height of 0.8 inches. Figure 5-8 shows that this resulted in a 3% reduction of power. Note that this is not the smallest power because the tube length, tube thickness and effectiveness were held constant. A more complete optimization would allow all parameters to vary. Also, note that a minimum is not shown with respect to the fin pitch, because, as stated in Section 3, the effect of the fin pitch on the F and J curves was neglected for this study.

Hence, a complete optimization would allow all parameters to vary and account for the geometric effects on the F and J curves. Therefore, future studies concerned with an actual design would follow the above suggestions in order to obtain an optimum dry cooling tower design.

MULTI-PORT TUBE GEOMETRIC PARAMETERS

Figure 5-1

OPTIMIZATION OF MULTI-PORT TUBES

d_H = HYDRAULIC DIAMETER

A = PORT FLOW AREA

PORT FILLET RADIUS $= r \geq 0$

Figure 5-2

COST AS A FUNCTION OF TUBE LENGTH AND TUBE THICKNESS

TUBE LENGTH AND TUBE THICKNESS OF MULTI-PORT TUBES
FOR MINIMUM TOTAL POWER

Figure 5-4

PERFORMANCE CHARACTERISTICS OF MULTI-PORT TUBES

Figure 5-5.

COMPARISON OF SINGLE-PORT AND MULTI-PORT TUBES

Figure 5-6A

COMPARISON OF SINGLE-PORT AND MULTI-PORT TUBES

Figure 5-6B

Figure 5-7

EFFECT OF FIN PITCH AND FIN HEIGHT ON MULTI-PORT TUBE PERFORMANCE

EFFECT OF FIN PITCH AND FIN THICKNESS ON MULTI-PORT TUBE PERFORMANCE

Figure 5-8

6.0 COOLING MODULE INSTALLATION CONSIDERATIONS

Because of the large heat rejection rate, forced or induced draft dry cooling towers must be constructed with a large number of fan cooling modules composed of one or more cooling coils. Each coil may be considered to be a separate air to water heat exchanger assembly comprised of a given number of fin tubes, tube sheets, closures, manifolds, etc. Installation of the cooling coils may be vertical, horizontal or arranged in deltas, such that one fan may service one or more coils. Although a detail study of the advantages, disadvantages and costs associated with all of these arrangements is beyond the scope of this study, a brief evaluation was made for several installation arrangements, in order to ascertain the advantages and limitations, if any, with the use of the C-W integral fin tubes installed in dry cooling modules. The results of these investigations are presented below.

6.1 Geometric Considerations

A number of design layout evaluations were made in order to evaluate installation arrangements with the C-W integral fin tube. Two basic arrangements were considered, i.e., the use of straight tubes and the use of bent or serpentine tubes. The latter was considered in order to evaluate the potential for reducing the number, and hence, the cost of tube joints. This would also result in the use of longer fin tubes which would further tend to reduce capital costs.

6.1.1 Straight Tube Geometries

A conceptual layout arrangement for the use of straight multi-port tubes in an induced draft dry cooling tower module was made as shown in Figure 6-1. This preliminary layout was made for comparative purposes with bent tube arrangements and to ascertain whether any unique advantages or disadvantages would occur with the use of multi-port tubes as compared to single port tubes. For a typical modular assembly using multi-port tubes (Figure 6-1), straight tubes are headered to carbon steel tubesheets with the water seal provided by a rubber O-ring which is maintained in position by a retainer plate as shown in Figure 2-5. A similar system is employed by C-W in continuous experimental equipment and provides an economical and practical tube to tubesheet

assembly, as well as providing an economical and practical transition between the aluminum tubes to the carbon steel tubesheet. The remaining assembly consists of carbon steel tubesheets, commercial type of pipe couplings and pipe manifolds. A probable type of manifolding arrangement for numerous cooling modules is shown in Figure 6-2. This schematic shows a manifold arrangement suitable for the above described modular installation of Figure 6-1. Use of single port or round tubes in the above installation would differ only in that the round tube ends would be expanded and flared into drilled holes in the tubesheet. One major cost disadvantage occurring with round tubes is that aluminum tubesheets would be required with the use of aluminum tubes.

6.1.2 Bent Tube Geometries

A conceptual layout employing multi-port tubes in a serpentine arrangement is shown in Figure 6-3. In this configuration, a single tube is bent to 180° sections such that the fin tips of consecutive bends are just touching to prevent air bypassing the fin tips. Since the mean tube bend radius is equal to the fin height, a maximum tube thickness is established by the tube physical properties and the fin height. For the purposes of this study, this thickness has been established at approximately 0.7 inches for a 1.0 inch fin height. An aluminum or carbon steel frame can be used to provide the structural rigidity and the mounting provisions for this type of core assembly. This serpentine geometry currently is commercially employed on a production basis with this type of multi-port tubes in airconditioning condensers on a number of U.S. automobiles. A major advantage of this type of geometry in dry cooling cores is the use of only two tube joints per cooling coil. Since the number of tube joints is inversely proportional to the tube length, it is advantageous to consider the use of the maximum potential tube length that may be available for this application. Since this type of tubing length is only restricted by the floor length available, an arbitrary maximum tube length of 200 feet was considered for comparative evaluation purposes in this study.

Recognizing the potential increase in tube side pressure loss and drain time limitations with tube lengths of 200 feet, a second bent tube arrangement was evaluated. This configuration, shown in Figure 6-4, employs a series of

serpentine tubes which are assembled in a stacked core arrangement. This geometry provides two potential advantages compared to the geometry of Figure 6-3. The first is that a greater vertical core height is obtainable for a given tube length and a fixed horizontal length, and secondly, thicker tubes may be employed, since the tube bend radius is considerably larger than the allowable tube bend radius of Figure 6-3. To prevent bypass air at the fin tips of the minimum bend radius tube in this configuration, a blank-off plate must be employed. Similarly, suitable blank-off plates are employed at the outer periphery of the tube bundles; these plates are made as a part of the structural framework to contain and support this tube core assembly. These flow blockage areas were also considered in the performance evaluations of this geometry. An alternate heading arrangement with this configuration provides for the inlet and outlet manifolds to be located on opposite sides of the cooling coil.

Figure 6-5 shows a probable arrangement of cooling modules employing either of the above bent tube configurations. In this arrangement, the cooling coils are assembled in a back to back assembly with inlet and outlet fittings manifolded on the same side of each coil. This arrangement is, therefore, fully compatible with either bent tube geometry, i.e., Figure 6-3 or 6-4. Similarly, the installation arrangement of Figure 6-5 is compatible with a straight tube geometry in which the coils are assembled in a back to back assembly with the water inlet and outlet fittings located on the same coil side such as might be used in a two pass system. Conversely, the installation arrangement of Figure 6-2 is also compatible with bent tube configurations in which the inlet and outlet fittings are installed on opposite sides of the coil. Therefore, it can be concluded that no unique manifolding advantages or disadvantages occur with cooling modules employing straight vs. bent tube core geometries. However, the cost advantages associated with the reduced number of tube joints still applies to the bent tube core geometries.

6.2 Performance Considerations

6.2.1 Tube Side Pressure Loss

An evaluation was made to determine performance advantages, if any, with bent tube geometries compared to straight tube geometries. Here it is noted that

the air side thermodynamic performance is the same for both configurations since the air face mass velocity is assumed to be the same in both geometries. However, the water side thermodynamic performance is affected by the cooling coil core configuration because of variations of tubing length, hydraulic diameter and flow area. This statement is illustrated by Figure 6-6 which uses the example of a 200 foot long multi-port tube in a serpentine geometry. As shown by Figure 6-6, in order to decrease the water pressure drop, a shorter tube, or a larger tube thickness, or a lower velocity must be used. However, increasing the tube thickness or decreasing the water velocity reduces the value of the water heat transfer coefficient. But, as shown by Figure 6-6, the overall heat transfer coefficient increases as the tube thickness increases and only decreases slightly as the water velocity decreases. Therefore, for a given length, a larger tube thickness will be superior to a smaller tube thickness so that cooling coils with tubes having a tube thickness less than 0.4 inches were not considered in this analysis. The results in Figure 6-6 were obtained with a serpentine tube length of 200 feet, having sixteen bends of 180 degrees and a horizontal length of 12 feet. The resulting tube side pressure loss due to the tube bends was found to be small, i.e., less than 3% of the total water losses for a .40 inch thick tube to less than 7% for a 1.0 inch tube thickness. Therefore, the use of serpentine tubes are not prohibited due to excessive tube side pressure losses resulting from the tube bending.

6.2.2 Tube Side Drain Time

The basic equation delineating drain time constraints was defined in Section 2.3. In particular, the effect head (H_e) was defined as the ratio of the vertical tube height to the total tube length. For the general case this can be defined as $H_e = [(L_Y + L_L)/L_T] \sin \delta$. Where δ is the angle of inclination from the horizontal and L_L is a vertical draining length of external core manifold having a flow area equal to the tube area. Numerical solution of this equation shows that the 90 second drain time requirement can always be achieved with reasonable values of L_L when δ is greater than zero. With $L_L = 0$, and the cooling cores arranged in a delta configuration with $\delta = 45^\circ$, like Figure 6-3, then the required drain time of 90 seconds can be met in a serpentine configuration only for the geometric values shown in Figure 6-7.

With multiple tube coil arrangements consisting of 2 or more tubes, the drain time of 90 seconds is more readily obtained (Figure 6-7). These results show that bent tube geometries can be constructed to satisfy the drain time requirement of 90 seconds. These results further show that although a performance advantage with a bent tube geometry is not available, the potential cost advantages of reduced number of tube joints, reduced tube handling and modular construction can be achieved with the use of bent tube core geometries similar to Figures 6-3 and 6-4.

Figure 6-1 Dry Cooling Tower Module with Straight Multi-Port Tubes

Figure 6-2 Manifold Arrangement for Straight Tube Cooling Modules

Figure 6-3 Dry Cooling Tower Module with Serpentine Multi-Port Tubes

**Dry Cooling Module with Parallel Arrangement
of Bent Multi-Port Tubes**

Figure 6-4

Figure 6-5 Manifolding Arrangement for Bent Tube Cooling Modules

MULTI-PORT TUBE PERFORMANCE IN SERPENTINE COIL CONFIGURATION

h_1 = WATER HEAT TRANSFER COEFFICIENT (BTU/HR FT² °F)

Figure 6-6

SERPENTINE DIMENSIONS

Figure 6-7

7.0 RELATIVE COST AND PERFORMANCE EVALUATIONS

A comparative evaluation was made to establish cost and performance advantages of the C-W integral fin tubes compared to existing fin tube geometries when used in dry cooling tower applications. Only the multi-port tube was considered in this comparison, since it was clearly shown in Section 5.0, that the multi-port tube is superior to the single port tube. In order to perform this comparative evaluation, cost estimates of currently available fin tube geometries were obtained from ERDA/PNL, Reference 9. These data were based on round tube geometries having continuous extruded or rolled type aluminum fins, and single and double footed wrap-on aluminum fins, both with 10 fins/inch. Tube costs were provided with both steel and aluminum liners having tube diameters of .50 to 1.25 inches. The comparative cost evaluation of this study was based on the cost of tube fabrication, cost of air and water pumping, cost of tube joints and fan system costs associated with the number and size of fans, inlet bells, gear boxes and motor drives. Comparative cost results for straight tube cooling coil geometries are given below:

7.1 Comparison of Tube Costs

In order to compare tube fabrication costs, an evaluation of the tubing requirements was made for each of the above finned tube geometries for the entire range of available round tube diameters, i.e., .50 to 1.25 inches. To obtain round fin tube costs and performance, only the maximum available fin height, i.e., maximum fin area, and therefore, least cost per surface area, was considered. These fin heights ranged from .25 inches at .50 inch tube diameter to .625 inch fin height at 1.25 inch tube diameter. In addition, the round fin tube performance was based on f and j characteristics obtained from Figure 10-77 of Reference 1.

Tube and fin costs for the multi-port tubes were calculated from the material costs and the fin machining costs. The tube and fin material costs were based on commercially available aluminum extrusion costs. Since the fin machining is performed on a modified punch press, the fin machining costs were based on labor rates associated with typical heat exchanger manufacturers. Consideration for normal inefficiencies, scrap, handling, etc., consistent with current manufacturing practice for this tubing, were assumed.

Finally, the fin machining costs were based on a constant machine rotating speed for each fin height and for all fin densities. Tooling costs were not considered for either the round fin tubes or the multi-port fin tube geometries in this study.

A comparison of the total amount and cost of fin tubing, the number and cost of fin tube joints, and the total pumping power, i.e., air plus water horsepower, was made for each of the above fin tube geometries. For comparative evaluation purposes, the multi-port tubes were compared at overall tube thickness equal to the round tube liner diameters. In this evaluation, the multi-port tube wall thicknesses are consistent with current extrusion and machining practice and consistent with available aluminum physical properties. A sketch of the multi-port fin tube geometries used in the evaluation of this section is given in Figure 7-1. For this comparison, the selected average cycle requirements of Section 3.0 were used to establish the total tube requirements. Also, this comparison was based on the same air flow per unit face area for all tube geometries. The optimum straight tube length for each tube geometry was selected at the minimum total pumping power. Although all of the round tube geometries are based on 10 fins per inch, this evaluation for the multi-port tubes was also made at 8 and 12 fins per inch. Comparative results are shown in Figures 7-2 thru 7-8 for 10 fins per inch, but the additional advantages with 8 or 10 fins per inch are tabulated at the end of this section. It should be noted that an optimum tube length was established at minimum total pumping power for each fin density, fin height and tube thickness. Therefore, the number of tubes, the tube costs, and the number and costs of tube joints all changed with fin density as recorded in the table at the end of this section. Based on results reported in Section 5.0, this cost evaluation for multi-port tubes was made using the .80 inch fin height as a base of reference. The effect of using 1.0 inch fin height is also shown in the following cost comparisons. Unlike the round fin tubes, the fin height of multi-port integral fin tubes does not depend on the tube thickness, but only depends on the tube material and the unique manufacturing process.

Cost savings for the use of multi-port tubes compared to the use of the above round fin tubes are shown in Figure 7-2 for 10 fins per inch. These results

show that substantial cost savings are available with the use of the multi-port tubes. Since the round tube and multi-port tube costs were relatively constant with tube diameter and tube thickness, respectively, an optimum tube diameter or tube thickness was not established. Also shown, is a small increase in multi-port tube costs using 1.0 inch fin height vs. .80 fin height. Additional evaluation of fin densities for multi-port tubes results in a 10.5% increase in multi-port tube costs for 8 fins per inch at 0.50 inch tube thickness to 12.0% at 1.25 inch thickness and .80 inch fin height. Similarly, a 7.5% decrease in multi-port tube costs was obtained for 12 fins per inch at 0.50 inch tube thickness to 8.5% decrease at 1.25 inch thickness. The effect of fin density on tube costs, operating costs and number of tube joints is summarized in the table at the end of this section.

7.2 Comparison of Operating Costs

Operating costs based on total air plus water pumping costs are compared in Figure 7-3. These results show that substantial cost savings are available with the use of the multi-port tubes. Also shown, is a small increase in potential savings with the use of 1.0 inch fin height compared to 0.80 inch fin height. Further evaluation for the effect of fin densities is also tabulated below. Figure 7-3 shows that the total pumping cost savings improve significantly at the larger tube thicknesses. This occurs because the pumping costs of round fin tubes increases at a greater rate with tube diameters than does the rate of pumping costs increase with multi-port tube thickness. The optimum tube thickness for maximum cost savings with multi-port tubes could not be established since available round tube data was limited to 1.25 inches. More important, however, is the optimum tube thickness for minimum total multi-port costs, based on total pumping costs and capital costs. This evaluation is discussed below in Section 7.3.

7.3 Comparison of Other Cost Parameters

The results given above were based on straight tube lengths which were calculated vs. total pumping power for each tube diameter or tube thickness. Optimum straight tube lengths were selected at the minimum total power. These selected tube lengths are shown on Figure 7-4. These results show that a significant increase in optimum tube length is available with the use of the

multi-port tubes compared to round fin tubes. This occurs because of the relatively larger flow areas on the tube side of multi-port tubes due to the placement of individual ports. Also shown, is a small decrease, i.e., 0 to 2%, with the use of 1.0 inch fin height vs. 0.80 inch fin height with multi-port tubes. It should be noted that the optimum tube length shown in Figure 7-4 for each fixed value of tube thickness differs from the optimum tube thickness shown in Figure 5-4 for each fixed tube length. This difference is due to the method selected for tube optimization as described in Section 5.2. Although further studies are required to establish the true optimum tube design for minimum total costs, the relative cost differences established in this section illustrate the general magnitude of cost savings that are obtainable with the use of the multi-port integral fin tubes.

Since this evaluation is based on the same total face area for each tube geometry, the results of Figure 7-4 yield a significant decrease in the potential number of tube joints, and hence, the number of required cooling modules. Each of these considerations provide further capital cost savings in structures, installation, fans, gear boxes, motors, housings, etc. An evaluation of the number of tube joints was made and these results are shown in Figure 7-5. These results show a substantial decrease in the number of tube joints resulting with the use of the multi-port fin tubes compared to round fin tubes. The effect of fin density on the number of tube joints is summarized in the table at the end of this section.

A preliminary estimate of the cost advantages with multi-port tubes due to the number of tube joints is shown in Figure 7-6. This evaluation was based on the total number of tube joints established in Figure 7-5 and an assumption for the type of tube joint. Tube joints for the round tubes were based on expanded and flared tubes in a drilled tubesheet. Carbon steel tubesheets were assumed for the steel tube liners while aluminum tube sheets were assumed for the aluminum liners. The use of multi-head drilling equipment was assumed. Tube joints for the multi-port tubes were based on the design shown in Figure 2-5. This design uses a carbon steel tube sheet assembly comprised of three separately punched plates. The tube slots are assumed to be individually made. The middle plate of the tubesheet assembly provides the required

water seal and this plate has the slots broached to size. The broaching was assumed to occur separately for each slot of the seal plate. An estimate of the tubesheet machining and tube assembly was made for each type of tube joint and the results are shown in Figure 7-6. These results show a significant cost savings with the use of multi-port tubes.

The use of longer fin tubes, without loss of pumping power, provides a further cost saving in the fan system, comprised of fan, gear box, motors and inlet bells. These savings occur due to the use of larger diameter fans as is shown in Figure 7-7. These results were synthesized from fan, inlet bells and motor cost data provided in Reference 2, and from gear box cost data established by Curtiss-Wright. These results clearly show the cost advantages and the large decrease in the number of fan modules occurring with the use of the larger fan diameters. The results of Figure 7-7 were combined with those of Figure 7-4 to establish the fan system cost savings available with the use of the C-W multi-port tubes compared to round fin tubes. These results are shown in Figure 7-8. Although additional fan system cost savings are available with either the round tube or the multi-port tubes with tube lengths greater than the optimum tube lengths of Figure 7-4, these additional tube lengths will result in increased total power consumption, and hence, a further trade-off would be required to establish an optimum cost. This trade-off was not done in this limited study.

Further cost savings will also occur with the use of multi-port tubes due to the longer allowable tubes, i.e., without power penalty, in the reduced amount of cooling module structures, control system, manifolds, land and facility equipment. This occurs because the longer tubes allow the use of larger diameter fans, and hence, a reduction in the number of cooling modules. These results are shown in Figure 7-9. The cost savings due to these factors were not established in this initial study.

Although a complete optimization of the multi-port tubes based on capital and operating costs is beyond the scope of this initial study, an indication of the magnitude of the cost savings is given by the above data. Also, an indication of the effect of geometric parameters which affect this optimization is given by the data provided herein. For example, these results show that

the total capital tube costs are relatively independent of tube thickness and only slightly affected by fin height. However, the operating costs are related to the tube thickness with a minimum occurring at about 0.50 inch tube thickness. Since operating costs are probably the most cost dominant factor, it would appear that the optimum multi-port tube thickness would occur near this value. Further cost savings due to the optimum tube length increasing with tube thickness, and its effect on fan system costs, reduced number of tube joints, reduced number of cooling modules, etc., will increase the optimum, multi-port tube thickness to perhaps a value between 0.50 and 0.75 inches. Obtaining this optimum value requires considerably more investigation than was available in this limited study.

A summary of the effect of fin density variation on the above cost parameters is shown on page 7-8 below for the above multi-port tubes. This data is compared to the 10 fins per inch tubes described above.

7.4 Cost Evaluation With Serpentine Tube Geometries

The above discussion and data presented in this section pertain to the use of straight tubes. These results showed significant advantages with the use of the C-W multi-port integral fin tubes compared to commercially available round fin tubes. However, additional advantages in costs, ease of handling, transportation, etc., are available with the use of the multi-port fin tubes in a serpentine tube arrangement. A description of this type of geometry was provided in Sections 2.0 and 6.0 of this report. A brief comparison of the serpentine geometry with a straight tube geometry is given below to illustrate these advantages.

For this comparison, a multi-port tube 200 feet long was selected at minimum total power based on the results shown in Figure 5-6. This resulted in a tube having 1.1 inch thickness and 10 fins per inch. For comparison purposes, a commercially available straight aluminum tube with extruded fins, 1.0 inch tube diameter and 10 fins per inch was selected. Based on results of Figure 7-4, a straight tube length of 40 feet was also selected at minimum total pumping power. This length is also consistent with maximum road shipment limitations. Evaluations of the multi-port tube were made for bent tube or

serpentine configurations similar to Figures 6-3 and 6-4 having 1, 2, 4 or 6 parallel tube paths. For each of these geometries, the required fin side blockages due to minimum tube bend radius was calculated and its affect on pumping power was determined and included in the evaluation.

Results of this investigation showed a savings in tube costs of approximately \$1.5 million with the use of the serpentine multi-port tube geometries. Similarly, other cost savings resulted, including \$0.4 million for tube joints, \$1.6 to \$1.9 million for the core construction, and \$3,456 per day for total pumping power at 10 mill power costs. These results showed only a small difference in cost savings, i.e., less than 2% for power and less than 1% for construction for the various serpentine geometries, i.e., 1, 2, 4 or 6 parallel tube paths. Thus, the selection of parallel tube paths would be established by drain time considerations, i.e., Figure 6-7, or by considerations for handling, installation, shipping, etc.

These results with a serpentine tube geometry are presented only to show relative and potential savings. No attempt was made to optimize the tube length, or to investigate other tube thicknesses, etc. In general, the results presented above for straight tubes apply to the serpentine tubes, with small changes for tube side pressure losses due to bends, and cost savings due to reduced number of tube joints.

PERCENT CHANGE IN MULTI-PORF TUBE COST PARAMETERS
COMPARED TO DATA AT 10 FINS PER INCH

FIN HEIGHT, INCHES	1.0				0.8			
TUBE THICKNESS, INCHES	.5	.75	1.0	1.25	.5	.75	1.0	1.25
Tube Cost								
@ 8 Fins/In.	+9.0	+9.5	+10.0	+10.5	+10.5	+11.0	+11.6	+12.0
@ 12 Fins/In.	-6.5	-6.7	-7.0	-7.3	-7.5	-7.8	-8.1	-8.5
Pumping Cost								
@ 8 Fins/In.	-6.5	-7.4	-8.3	-9.1	-6.2	-7.2	-8.1	-8.9
@ 12 Fins/In.	+7.2	+8.1	+9.1	+10.1	+6.7	+7.8	+8.9	+9.9
Optimum Tube Length								
@ 8 Fins/In.	+7.4	+6.3	+5.6	+5.1	+7.5	+6.9	+5.9	+6.0
@ 12 Fins/In.	-5.1	-4.6	-4.1	-3.8	-5.8	-4.8	-4.3	-4.0
No. of Tube Joints								
@ 8 Fins/In.	-6.9	-6.0	-5.3	-4.9	-7.0	-6.4	-5.5	-5.7
@ 12 Fins/In.	+5.3	+4.8	+4.3	+4.0	+6.2	+5.0	+4.6	+4.1
*Fan System Cost								
@ 8 Fins/In.	-0.5	-0.5	-0.6	-0.7	-0.5	-0.6	-0.6	-0.8
@ 12 Fins/In.	+0.3	+0.4	+0.4	+0.5	+0.4	+0.4	+0.5	+0.5

*At Fan Diameters Consistent with Optimum Tube Length

NOTE: Optimum tube lengths were established at minimum total power for each fin density, fin height and tube thickness.

DESCRIPTION OF MULTI-PORT TUBE
USED IN COMPARATIVE ANALYSIS

Figure 7-1

TUBE COST SAVINGS USING
CW MULTI-PORT TUBES COMPARED TO ROUND FIN TUBES

ALL DATA AT 10 FINS PER INCH
ALL DATA WITH ALUMINUM FINS

Figure 7-2

OPERATING COST SAVINGS USING
CW MULTIPOST TUBES COMPARED TO ROUND FIN TUBES

ALL DATA TA 10 FINS PER INCH
ALL DATA WITH ALUMINUM FINS
POWER COST = 10 MILLS PER KW-HR

Figure 7-3

OPTIMUM TUBE LENGTHS USING
CW MULTIPOINT TUBES COMPARED TO ROUND FIN TUBES

ALL DATA AT 10 FINS PER INCH
ALL DATA WITH ALUMINUM FINNS

Figure 7-4

TOTAL NUMBER OF TUBE JOINTS USING
CW MULTIPOST TUBES COMPARED TO ROUND FIN TUBES

Figure 7-5

TUBE JOINT COST SAVINGS USING
CW MULTIPOINT TUBES COMPARED TO ROUND FIN TUBES

Figure 7-6

FAN SYSTEM COST VERSUS FAN DIAMETER

INCLUDES: FANS
GEAR BOXES
MOTORS
INLET BELLS

Figure 7-7

FAN SYSTEM COST SAVINGS USING
CW MULTI-PORT TUBES COMPARED TO ROUND FIN TUBES

ALL DATA AT 10 FINS PER INCH
ALL DATA WITH ALUMINUM FINS

Figure 7-8

NUMBER OF FANS REQUIRED USING
CW MULTI-PORT TUBES COMPARED TO ROUND FIN TUBES
FOR FAN DIAMETERS CONSISTENT WITH OPTIMUM TUBE LENGTHS

Figure 7-9

8.0 REFERENCES

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9.0 LIST OF SYMBOLS

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
A	Water flow area per port	ft ²
A _F	Total heat transfer area of the fins	ft ²
A _I	Total heat transfer area of the internal fluid passage	ft ²
A _O	Total outside heat transfer area of the fins and tube	ft ²
c _p	Specific heat	Btu/lb °F
d _H	Hydraulic diameter	ft
d _I	Internal diameter	ft
D _H	Multi-port tube water pressure drop parameter, equals $d_H A^2 / (S_T + H_W)^2$	ft
D _S	Single port tube water pressure drop parameter, equals $d_H A^2 / T_W^2$	ft
E	Heat exchanger effectiveness	None
f, F	Friction factor	None
F _H	Fin height	ft
F _L	Fin length	ft
F _T	fin thickness	ft
g	Acceleration due to gravity	ft/hr ²
G	Mass velocity	lb/hr ft ²
h _I	Water heat transfer coefficient	Btu/hr ft ² °F
h _O	Air heat transfer coefficient	Btu/hr ft ² °F
H _e	Effective head	None
H _H	Hole height	ft
H _W	Hole width	ft
j, J	Colburn j factor	None

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
k	Thermal conductivity	Btu/hr ft °F
K	Scaling constant used to convert electrical resistance to the shape factor, equals $S R_e$	Ohms
K_B	Bend water pressure drop coefficient	None
L_L	Length of drain leg	ft
L_T	Tube length	ft
L_X	Horizontal dimension of cooling coil	ft
L_Y	Vertical dimension of cooling coil	ft
N_B	Number of tube bends per tube length	None
N_i	Internal fin effectiveness	None
N_o	External fin effectiveness	None
N_P	Number of ports	None
P_a	Required fan power	H.P.
P_w	Required pump power	H.P.
Δp_a	Air pressure drop	Inch HgA
Δp_w	Water pressure drop	psia
q	Heat rejection rate	Btu/hr
q'	Heat rejection rate per foot of tube	Btu/hr ft
q_e	Electrical current flow rate	amps
r	Fillet radius of multi-port ports	ft
R	Overall heat transfer resistance	hr °F/Btu
R_e	Electrical resistance	Ohms
R_i	Water heat transfer resistance, equals $1/N_i h_i A_i$	hr °F/Btu
R_m	Metal heat transfer resistance, equals $1/k ST_L$	hr °F/Btu

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
R_o	Air heat transfer resistance, equals $1/N_o h_o A_o$	hr °F/Btu
S	Shape factor	None
S_T	Support thickness for single port tubes	ft
t	Time	sec
t_D	Draining time	sec
T_a	Air temperature	°F
T_T	Tube thickness	ft
T_w	Water temperature	°F
T_W	Tube width	ft
ΔT_m	Temperature drop through the tube metal	°F
U_o	Overall heat transfer coefficient	Btu/hr ft ² °F
V	Water velocity	ft/sec
V_Y	Vertical height for one return bend	ft
ΔV	Voltage difference	volts
W_a	Air flow rate	lb/hr
W_w	Water flow rate	lb/hr
X	Length of tube drained in time t	ft
α	Angle the cooling coil makes with the horizontal	degrees
ρ_a	Density of air	lb/ft ³
ρ_w	Density of water	lb/ft ³